Simulation of Adaptive Cruise Control for Control Engineering Education Purposes

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1 Background

Adaptive cruise control (ACC) for vehicles has been available for years now. It was created on the top of cruise control that originates roughly in the 1950's. The ACC is to enhance the automatic driving by keeping a safety distance set by a driver to another vehicle in front. The ACC is basically about balancing between the vehicle speed and the distance between two vehicles.

For control engineering education purposes, the ACC is a representative example of control engineering. Basically, anyone can understand the scope of the ACC requiring little effort on motivating the need for the ACC. Second, the ACC engineering task can be divided into well-defined, separate tasks of process modelling, control design and control tuning which all can be verified by applying dynamic, time-domain simulation.

The literature of the ACC recognizes several ways to deal with the ACC and the vehicle models to be worked upon. The reader is advised to get familiar with work by Bengtsson (2001), Gäfvert (2003), Kim (2012) and Miyata (2010). For gaining a good insight to process modelling and control design with tuning, the work by Vilanovia & Visioli (2012), Åström & Hägglund (1995) are recommended.

2 Aims

The scope of the paper is to design an ACC strategy based on a simple differential equation for modelling a vehicle. The model contains one manipulatable input, vehicle's engine thrust, and one putout, vehicle speed. In addition, the model recognizes two primary disturbance terms: road slope and air resistance.

The proposed model is then used for designing an advanced PID control for regulating both vehicle speed and the safety distance to the front vehicle. As there are two simultaneous control tasks (speed and distance) but basically only one variable to be manipulated, engine thrust, a single PID controller is not adequate. Instead, an advanced control scheme is required with overriding (limiter) control. Also, adaptive control is discussed as an addition to the introduced control strategy.

The emphasis in this paper is an educational perspective consisting of explaining the simplifications made in modelling, decisions made in control design and, finally, observations made in control tuning. Results of modelling, control design and tuning for the ACC are visualized through simulations executed in Matlab/Simulink environment.

3 Vehicle model

Quite often, designing a good control scheme does not necessarily require a perfect match in modelling. Instead, a simple model is adequate as far as it captures the essence of the behavior. Following this principle, a simple force-based model that was chosen for the ACC design purposes, is a kinematic first-order, non-linear Newtonian model. Applying a free body diagram as illustrated in figure 1, the model sums up affecting forces which are vehicle engine force F_{eng}, gravitational force G and force due to air resistance F_{air}.

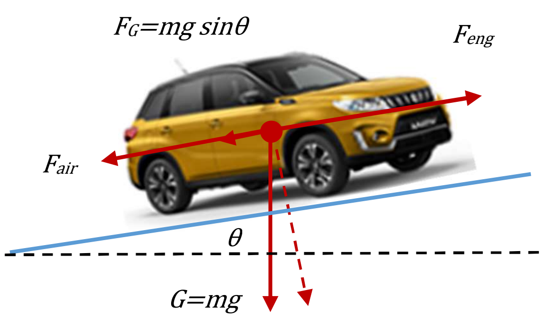

Figure 1. Free-body diagram of a vehicle (Suzuki Vitara 4WD 2019) with mass m on a road surface having a slope of θ.

According to the second Newtonian law, the sum of affecting forces applied to a vehicle generates acceleration $a(t)$ with respect to time t

$$m \cdot a(t) = F_{eng}(t) - F_G(t) - F_{air}(t) \quad (1)$$

By substituting forces $F_G(t) = mg \cdot sin\theta(t)$ and $F_{air}(t) = bv^2(t)$ and replacing acceleration by the first-order time derivative of speed $a(t) = \frac{d}{dt}v(t)$, the equation (1) can be expressed as

$$m \cdot \frac{d}{dt}v(t) = F_{eng}(t) - mg \cdot \sin\theta\,(t) - bv^2(t) \tag{2}$$

where m is vehicle mass with a driver (kg), v is vehicle speed (m/s), F_{eng} is engine thrust (Newton), θ is road slope (angle, radians), b is air resistance factor (kg/m) and $\frac{d}{dt}$ is a differential operator. For control purposes, the model output is the speed v and the manipulated input is the engine thrust F_{eng}. The road slope is θ considered as a load disturbance.

The effect of air resistance in the equation (1) lies on the negative sign of the force and in its parameter b which contains physical properties of the vehicle resisting its engine force. Also, the model (1) does not include the variable measuring the distance to the front vehicle. This is to be added later, and once completed, it brings in another load disturbance.

The equilibrium point of the non-linear differential equation (2) can be found by expressing the model as

$$v'(t) = \frac{1}{m}F_{eng}(t) - g\sin\theta\,(t) - \frac{b}{m}v^2(t) \tag{3}$$

In steady state, $v'(t) = 0$ equaling to a constant speed which results in an equilibrium condition

$$F_{eng}(t) = bv^2(t) + mg\sin\theta\,(t) \tag{4}$$

At any time, the thrust generated by a vehicle's engine depends on the speed and the slope of the road.

Figure 2 illustrates the required engine thrust for speed up to 120 km/h and for road slope up to 10 %. In this context, the road slopes are percentages of right angle, that is, 5 % corresponds to an angle of $5\,\% \cdot 90° = 1.8°$. The required engine force increases rapidly for increasing speed but linearly for increasing road slope.

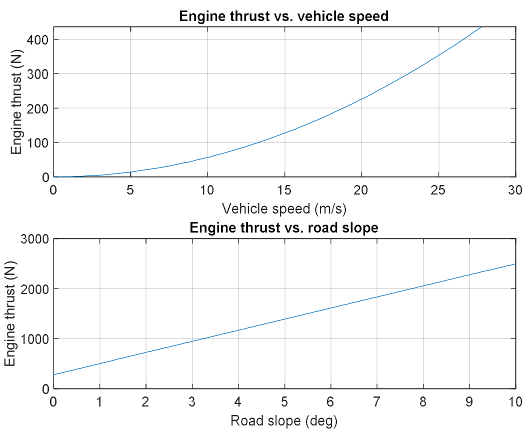

Figure 2. Required engine thrust of a vehicle to maintain a constant speed for speeds and road slopes.

4 Linearization of vehicle model

For simulation, the non-linear model such as (2) or (3) could be used. However, control design gets much simpler if the model is linear. In addition, the linear model is more comprehensible providing with insight to the model properties such as dynamics and gains.

For linearization, the non-linear differential equation is formulated as

$$f(t) = mv'(t) - F_{eng}(t) + mg\sin\theta\,(t) + bv^2(t) = 0 \tag{5}$$

The sine term $\sin\theta\,(t)$ can be eliminated in the equation by observing that for small values of $\theta(t)$ the sine terms equals to a slope itself: $\sin\theta\,(t) \approx \theta(t)$.

Now, the equation (5) is simplified to

$$f(t) = mv'(t) - F_{eng}(t) + mg\,\theta\,(t) + bv^2(t) = 0 \tag{6}$$

For linearization, the first order Taylor series expansion of the function (6) is

$$df(t) \approx \Delta f(t) = \frac{\partial f}{\partial v'(t)}\Delta v'(t) + \frac{\partial f}{\partial F_{eng}(t)}\Delta F_{eng}(t) + \frac{\partial f}{\partial \theta(t)}\Delta\theta(t) + \frac{\partial f}{\partial v(t)}\Delta v(t) \tag{7}$$

where Δ indicates small differences from the nominal, linear values of variables. After applying the Taylor series expansion, the linearized function is

$$\Delta f(t) = m\Delta v'(t) - \Delta F_{eng}(t) + mg\Delta\theta(t) + 2bv_0\Delta v(t) = 0 \tag{8}$$

where v_0 is speed used for linearization.

The resulted equation is a first-order ordinary differential equation of speed. The equation could be solved explicitly for $\Delta v(t)$ but that is not necessary as the equation can be Laplace-transformed to a transfer function which is much more applicable for control design and simulation.

5 Transfer function models

For both control design and gaining a better understanding of the vehicle model, the linearized model (8) is given in Laplace-domain as

$$m\big(s \cdot \Delta v(s) - \Delta v(0)\big) + 2bv_0\Delta v(s) - \Delta F_{eng}(s) + mg\Delta\theta(s) = 0 \tag{9}$$

where s is a Laplace-domain variable replacing time t. By assuming the speed difference being to zero in zero time, that Is, $\Delta v(0) = 0$, the equation is final:

$$ms\Delta v(s) + 2bv_0\Delta v(s) - \Delta F_{eng}(s) + mg\Delta\theta(s) = 0 \tag{10}$$

Now, by arranging the terms in (10), the Laplace equation is given as

$$\Delta v(s) = \frac{1}{ms+2bv_0}\Delta F_{eng}(s) - \frac{mg}{ms+2bv_0}\Delta\theta(s) \quad (11)$$

There are two input variables, engine thrust ΔF_{eng} and road slope $\Delta\theta(s)$, and one output variable, speed Δv. There are two transfer functions $P_{eng}(s)$ and $P_\theta(s)$, one for each input variable

$$\Delta v(s) = P_{eng}(s)\cdot\Delta F_{eng}(s) + P_\theta(s)\cdot\Delta\theta(s) \quad (12)$$

where the transfer functions are

$$P_{eng}(s) = \frac{\Delta v(s)}{\Delta F_{eng}(s)} = \frac{\frac{1}{2bv_0}}{\frac{m}{2bv_0}s+1} = \frac{k_{eng}}{\tau s+1} \quad (13a)$$

$$P_\theta(s) = \frac{\Delta v(s)}{\Delta\theta(s)} = -\frac{\frac{mg}{2bv_0}}{\frac{m}{2bv_0}s+1} = \frac{k_\theta}{\tau s+1} \quad (13b)$$

Both transfer functions (13a, 13b) have static gains $k_{eng} = \frac{1}{2bv_0}$ and $k_\theta = -\frac{mg}{2bv_0}$ indicating how much speed is eventually changed if one of the inputs is changed in a stepwise manner by a certain amount. The units for the gains are $\frac{\frac{m}{s}}{N}$ (engine) and $\frac{\frac{m}{s}}{rad}$ (road slope). The static gains have opposite signs as they have opposite impacts on speed: a positive change in engine thrust causes a positive speed change whereas a positive chain in road slope (going uphill) causes a negative change in speed.

For example, if the nominal (linearized) speed is $v_0 = 80\ \frac{km}{h} = \frac{80}{3.6}\ \frac{m}{s}$, air resistance factor $b = 0.57\ \frac{kg}{m}$ and vehicle mass 1220 kg with a 80 kg driver is $m = 1300$ kg (as per Suzuki Vitara 4WD 2019) and gravity constant $g = 9.82\ m/s^2$, then the static gains are $k_{eng} \approx 0.039\ \frac{\frac{m}{s}}{N}$ and $k_\theta \approx -504\ \frac{\frac{m}{s}}{rad}$. Now, if the engine thrust is increased rapidly by 10 N, the speed is increased finally by 0.39 m/s if the road slope does not change at the same time. Similarly, if the road slope is changed only by 1/10 ° ($\approx 0{,}00175\ rad$), the speed is decreased eventually by 0.88 m/s.

Both transfer functions (13a, 13b) are stable as they both have a single, real-valued negative pole $p = -\frac{2bv_0}{m}$. Consequently, the system output speed settles to a bounded value if an input (engine thrust or road slope) has a bounded change.

The transfer functions have time constant $\tau = \frac{m}{2bv_0}$ in common. The unit for the time constant is second. The time constant is an indicator of the speed of the system: the smaller the time constant, the faster the system is. For the values (Suzuki Vitara) given before, the time constant for a vehicle would be $\tau \approx 52\ s$. For a rapid stepwise change in road slope or engine thrust, appr. 63 % of the final speed change is achieved in 52 seconds.

6 Model parameters

The transfer functions contain four parameters: vehicle mass with a driver, air resistance factor, nominal speed and gravitational constant. As the last parameter is rather constant, the truly interesting parameters affecting the model are mass, air resistance and nominal speed. Table 1 shows the relationships between the primary model parameters (mass, air resistance, nominal speed) and transfer function parameters (static gain and time constant).

Table 1. Impact table of model parameters (mass, air resistance, nominal speed) affecting model properties (time constant and static gains).

Model parameters	Time constant τ	Static gain k_{eng} (absolute value)	Static gain k_θ (absolute value)
Vehicle mass m	Increases	No impact	Increases
Air resistance b	Decreases	Decreases	Decreases
Nominal speed v_0	Decreases	Decreases	Decreases

Table 1 clearly indicates that if the vehicle mass is increased by having more load or passengers, the time constant increases making the vehicle as a system slower. Then, a change in road slope or engine thrust results in a speed change which is slower than for a smaller mass. An increased vehicle mass increases also the static gain for a road slope model P_θ but, surprisingly, it has no impact on the static gain for an engine model P_{eng}.

Increasing air resistance and nominal speed cause a decrease in time constant for both systems and for both static gains making both models P_{eng} and P_θ faster and more reagent. The only difference in causality comes from the opposite sign of the static gains k_{eng} and k_θ.

In literature, air friction coefficient is typically expressed as a function of air density ρ, cross-sectional area A and drag coefficient C_d:

$$b = \frac{1}{2}\rho A\cdot C_d \quad (14)$$

Air density is not constant but varies according to air temperature and pressure. For simplicity, the air density is assumed to be $\rho = 1.20\frac{kg}{m^3}$ which applies for $+20$ °C air temperature and 1 bar atmosphere pressure. The cross-section area of a vehicle is assumed to be $A \approx 2.86\ m^2$ (as per Suzuki Vitara 4WD 2019 with width 1775 mm and height 1610 mm) and $C_d = 0.33$. With the given values, the air resistance factor is $b \approx 0.57$.

Figure 3 shows open loop step responses for linearized models (13a, 13b). In simulations, the engine thrust has been increased by 10 N showing the resulted speed change with respect to time. Similarly, the road slope has been increased by 1/10 ° showing its impact on the speed, too. The simulations are independent showing the speed dynamics when only one of the input variables, engine thrust or road slope, is changed at a time another remaining unchanged.

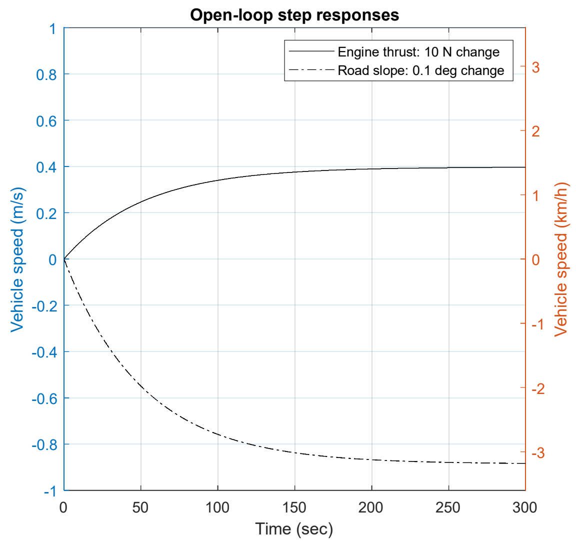

Figure 3. Open-loop step responses for linearized transfer function models (13a, 13b). Upper: speed vs. engine thrust using model P_{eng}. Lower: speed vs. road slope using model P_θ.

The mass of the vehicle m affects both time constant τ and static gain k_θ of the road slope model P_θ. The mass is variant as it includes not only the vehicle but also a driver with possible passengers. At any time, the real mass depends on passengers. Table 2 shows how much the time constant alone is affected as a function of passengers. It shows that the change can be about +25 % at the maximum. This observation gives rise to consideration of robustness of control design to guarantee closed-loop robustness and stability. Furthermore, adaptive control strategy could be considered.

Table 2. Impact of number of passengers on vehicle dynamics (time constant).

Nr of passengers	Total mass (kg)	Time constant τ (sec)	Time constant change (%)
0	1300	51.7	0
1	1380	54.9	+6.2 %
2	1460	58.1	+12.3 %
3	1540	61.2	+15.5 %
4	1620	64.4	+24.6 %

7 Control strategy for ACC

Vehicle or ACC manufactures typically implement their ACC control schemes as a black-box realization with no transparency to vehicle owners and drivers. User manuals have very little information available on the details of the applied ACC control strategy. However, operability of the ACC with available driver-specific settings is well instructed in manuals.

For control engineering education, the ACC leaves several options for designing a control strategy. Each strategy has its benefits and pitfalls. In this paper, the proposed PID control design for ACC incorporates two PID controllers with overriding control features.

The control objective of the ACC is to secure safe driving by limiting the distance to a vehicle in front driving to the same direction. Another objective is to provide with a good target speed regulation by attenuating load disturbances such as road slope and air resistance changes. The third objective is to have a smooth setpoint response for a changed speed setpoint.

The selected control strategy involves two feedback controllers. The primary controller is for regulating the vehicle speed as set by a driver. There is an inbuilt speed sensor in a vehicle providing with a real-time measurement of the vehicle speed. The speed controller (PI controller) reacts on the target speed and the measured speed.

Another PI controller is for keeping the safety distance to a vehicle driving in front. There is a radar measuring the distance between the cars and the controller reacts on the measured distance comparing it to the user-selected safety distance which is the setpoint for the safety distance controller. The usage of two controllers for manipulating one variable is called limiting control or overriding control.

Table 3. ACC strategy in terms of signals and features.

Signal or feature	Speed control	Safety distance control
Setpoint	Speed set by driver	Distance set by driver
Measurement	Speed sensor	Radar
Control output	Engine thrust	Engine thrust
Controller type	Proportional-Integral	Proportional-Integral
Control strategy	Limiter control (minimum selection)	

The derived, linear transfer function model (12) does not include a safety distance as an input or a parameter. However, the safety distance between the vehicle and another vehicle in front is a significant variable which requires attention. The safety distance is typically measured using a scanning radar mounted on the vehicle's front. The radar returns a distance to a vehicle in front. The distance is an important measurement that can be used as a safety measure for implementing adaptive cruise control (ACC). For simplicity, it is assumed that the measurement is well calibrated returning precise measurements. Table 3 collects the essence of the selected ACC strategy in terms of signals and control features.

Figure 4 illustrates the ACC strategy with two controllers: speed and distance controller. Both controllers have their setpoints and measurements for speed and distance to a vehicle driving in front. The controllers manipulate the vehicle's engine thrust but only one of the controllers is permitted to control at a time. There is a minimum selector for selecting the smaller of two control signals. This is to guarantee that the speed controller may not increase the speed if there is a vehicle inside the safety distance, that is, safety distance setpoint for the distance controller. The minimum selected control is taken to the engine causing speed and distance to a vehicle in front which are measured and fed back to the controllers.

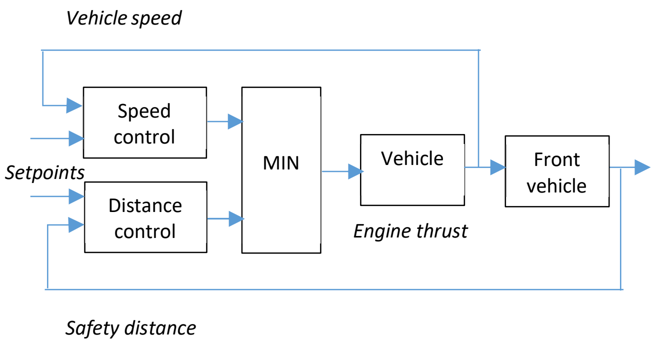

Figure 4. Block diagram of ACC control strategy.

8 Controller tuning

As the open-loop dynamics involves only first-order dynamics, a standard PI controller (Proportional-Integral) with only two tuning parameters k_p (proportional gain) and t_i (integral time) is adequate for control. The PI controller is of the ISA standard from

$$u(t) = k_p\left(e(t) + \frac{1}{t_i}\int_0^t e(\tau)d\tau\right) \tag{15}$$

where u is the PI controller output for regulating the vehicle's engine thrust $u(t) = F_{eng}(t)$ and e is a control error $e(t) = r(t) - y(t)$.between the target speed r and the real-time speed measurement y.

The control objectives are smooth speed setpoint following and good load disturbance attenuation with adequate robustness to model uncertainties. As the open-loop dynamics of the model between the engine force and the speed involves no dead time in practice, the PI controller can be tuned to

$$k_p = \frac{1}{k_{eng}}\frac{\tau}{\tau_{set}} \tag{16a}$$

$$t_i = \min\left(\tau, 4\tau_{set}\right) \tag{16b}$$

where the tuning guidelines follow the SIMC tuning method by Skogestad (Vilanova & Visioli, chapter by Skogestad & Grimmholt, 2012) with closed-loop time constant τ_{set} being the only design parameter for an engineer to choose. By selecting $\tau_{set} < \tau$, the PI controlled feedback vehicle is targeted to be faster than in open-loop with no speed controller. The tuning method is known to provide tunings with robustness to model uncertainties.

Figure 5 plots setpoint and load disturbance responses for a PI controlled speed for different PI tunings with . The upper-left corner shows setpoint responses when a driver has changed a setpoint target by +1 m/s (ca. 3.6 km/h) at time $t = 0$. As the nominal speed for a linear model is $v_0 = 80$ k/m, the actual speed target is ca. 83.6 km/h being 80 km/h before the change. The lower-left corner shows the controller output (engine thrust) required on top of the appr. 281 N which is required to maintain speed of 80 km/h before the change. The maximum required engine thrust change immediately after the setpoint change is almost 100 N but the final thrust change is only ca. 25.4 N for keeping the new speed target.

The upper-right corner shows load disturbance responses when a road slope has changed by 1 ° at time $t = 0$. The speed target remains unchanged as 80 km/h but the road slope change causes a minor decrease in speed which is at worst less than -0.02 m/s (ca. 0.07 km/h) and most probably goes unnoticed by a driver. The lower-right corner shows the controller output (engine thrust) on top of 281 N.

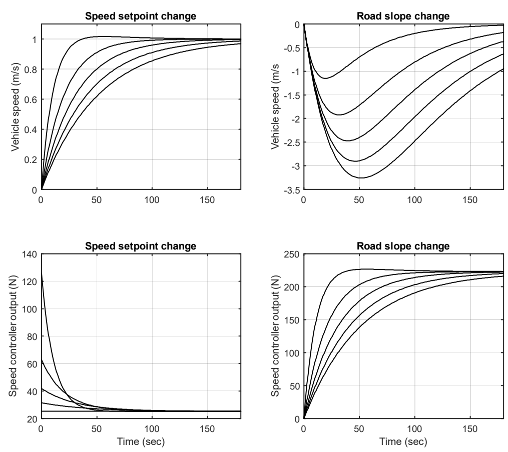

Figure 5. Closed-loop step responses for a PI controlled vehicle speed. Left: vehicle speed setpoint response (upper) and controlled engine thrust (lower). Right: vehicle speed load response (upper) and controlled engine thrust (lower).

9 Simulation results

The final simulation illustrated in figure 6 involves simulation of the feedback system as shown in figure 4. The speed PI controller was tuned to $\tau_{set} = 0.6 * T = 31.2$ giving $k_p = 42$ and $t_i = T = 52$ whereas the distance PI controller was tuned to be more aggressive by setting $k_p = 42$ and $t_i = 26$. The target speed initially is zero corresponding 22.2 m/s (80 km/h). At time of 30 seconds, a driver changes the target speed by 3 m/s (ca. 10.8 km/h). The ACC control strategy increases engine thrust to reach a new setpoint. At the same there is another vehicle in front which originally was three meters more far than a preset safety distance. The speed change takes the vehicle in front close, just about at the safety distance level which is set to zero in simulation.

This change is not adequate for the distance controller to take over. However, the car in front starts driving a bit slower making the safety distance smaller than the preset margin (set at zero in simulation) causing the distance PI controller to decrease its output. And it does not take long when its output is smaller than the output of the speed controller making it the controller in charge through the minimum selector as in figure 4. Finally, the speed is adaptively set to a lower level (app. -10 km/h) which is exactly the speed of the car driving in front making the safety distance be exactly at the pre-set level.

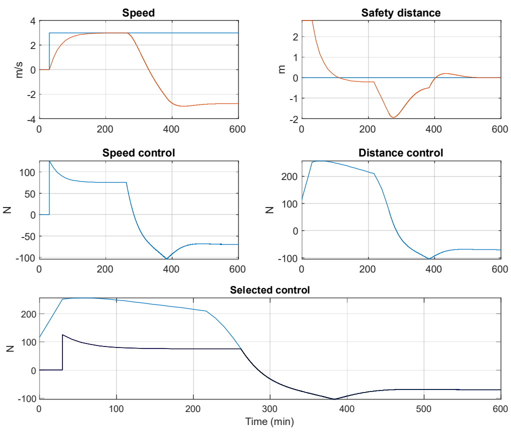

Figure 6. Simulation of ACC control strategy.

10 Discussion on adaptive control

The ACC adapts to a changing distance between two vehicles. However, it does not adapt to changes in vehicle dynamics. As shown in table 3, there can be a relative change of +25 % in time constant if the number of passengers is changed from zero to four. That changes open-loop dynamics remarkably and gives rise to discussion on adaptive control features to improve the ACC control strategy's robustness to altering dynamics. Any PI controller should be tuned quite sluggishly to provide with a such a stability margin (maximum sensitivity or gain/phase margins) that it would cope with any number of passengers and still provide with stabilizing control. And sluggishness is far from expectations of any driver driving a vehicle under cruise control.

Adaption would require recognition of vehicle mass. Most probably, the easiest way to do that would be sensing if a seat is carrying a passenger or any load in general. This would require only a few mass gauges, one for each seat and one for a trunk.

As the nominal speed v_0 is also a parameter affecting the open-loop dynamics, and, hence, the controller tuning, it could be similarly taken into consideration in an adaptive manner. Dealing with a nominal speed would not require additional sensors as the speed is already being measured as a primary variable.

11 Conclusion

The resulted feedback simulation model of the ACC can be used for control engineering education. More importantly, the whole engineering process of vehicle modelling, control design and tuning with simulation can be used as a case example for control engineering education at different stages of education level. Starting with modelling, the complexity of control engineering can be increased by introducing control design objectives and, finally, control design and tuning. Obviously, the simulation model also gives rise to design and simulate other control strategies than the one given by the author, not to mention, adaption mechanisms discussed in the previous section that could be applied to enhance the control performance.

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